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A study on the rotary steam engine for distributed generation in small size power plants

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ABSTRACT

The widespread use of the renewable energies requires compulsorily the development of new technological solutions for the efficient employment of these resources, such as small size, cogenerative power plants.

A small volumetric steam engine may be considered an interesting device for an efficient and flexible use the heat generated by biomasses combustion or solar thermal collectors. In effects, this kind of engine may be operated with different kind of fluids, is quite insensitive to high degrees of humidity at the end of the expansion and is able to manage small volumetric flow rates with no losses in conversion efficiency.

This paper shows the development of a rotary steam engine derived from a Wankel internal combustion engine. This engine was taken into account for this use due to its low bulk, vibration and noise running. Moreover, its use as an external combustion engine overrides its typical disadvantages that mainly belong to the combustion phase.

The numerical modeling and the experimental tests using compressed air in a first prototype are shown in this paper. The results of the experiments allowed the validation of the model developed and this was employed to carry out a first optimization of the engine by means of the increase of the discharge coefficient of the exhaust valves.

1. Introduction

Although the use of renewable energies such as biomasses is widely considered as one of the solutions for the arising troubles of pollutions, green house effect and energy resources diversification, due to their very widespread availability it is seldom convenient to concentrate the generation of electricity into a small number of large power stations due to the practical unfeasibility of the primary energy transportation, as underlined in the literature [1,2].

The use of the water steam Rankine cycle is not well established for very small size plants and the energy production from low-temperature heat still has some difficulties as well [3]. Organic working fluids may be taken into account [3–5] to simplify the plant layout, as well as ammonia, benzene or carbon dioxide [4,7], where in this last case a transcritical Rankine cycle must be employed. In this last case quite promising results were shown but its real application is limited both by the larger heat exchangers area and by the very high fluid pressure needed [7]. Thermal efficiencies ranging between 8% and 20% were reported [5,8] for the examined cases.

In this work the use of volumetric rotary engines was proposed upon the following considerations:

- Volumetric engines durability and reliability are almost insensitive even to quite high degrees of humidity [9] and they may be operated with a small fluid mass flow regardless of the pressure drop available [6,9].
- Rotary engines are able to rotate at quite high speed in a very silent way and with a very low degree of vibrations [10–12].
- They may be used with different fluids and volumetric flow rates by adjusting the rotating speed or the introduction degree.

Though significant technological improvements were reached in the past, the Wankel internal combustion engine never reached a wide volume of sales due to its well known combustion and lubrication inconvenients [10–15] that may be practically eliminated if it is used as an external combustion engine. Several schemes were proposed to realize volumetric rotary engines [16,17], but the Wankel mechanism was reckoned as the most promising technological solution [17–19]. Nevertheless, for this machine to be commercially successful, existing designs need to be modified and optimized.

In this work the analysis of the use of volumetric engines in very small size plants (in the range of 5–50 kW) was carried out. Though many working fluids may be suitable for this kind of applications,

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Nomenclature

e	introduction degree
γ	recompression degree
V	displacement of each chamber of the engine (m^3)
V_1	volume of the cylinder at the beginning of the intake phase (m^3)
V_2	volume of the cylinder at the end of the intake phase (m^3)
V_3	volume of the cylinder at the end of the expansion phase (m^3)
$V_{5'}$	volume of the cylinder at the beginning of the recompression phase (m^3)
V_5	volume of the cylinder at the end of the recompression phase (m^3)
α	angular rotation of the shaft
n	compression and expansion polytropic index

in this first analysis only the use of steam was considered. A numerical and experimental analysis was carried out by using a prototype built on the basis of a gasoline powered Ficht & Sachs Wankel engine.

2. The steam Wankel engine operation

The thermodynamic cycle on the pressure–volume diagram (p–v thermodynamic diagram) and the phases of a steam Wankel engine are respectively depicted in Figs. 1 and 2 and may be summarized in: steam introduction (1–2), expansion (2–3), exhausting (3–4, 4–5′), recompression (5′–1′). Then a new cycle begins when the rotor is faced to the other site of the case.

For a volumetric engine the introduction and recompression degrees may be defined as follows:

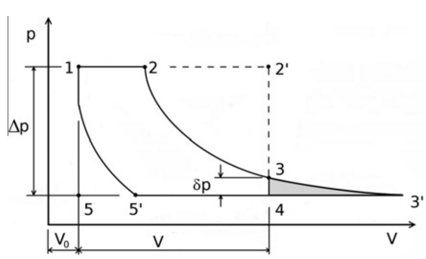

Fig. 1. Representation on the p–v diagram of a volumetric steam engine cycle.

$$e = \frac{V_2 - V_1}{V} \qquad \gamma = \frac{V_{5'} - V_5}{V}$$

This work was carried out considering fixed introduction and recompression degrees, that were chosen as a compromise between delivered work and thermal efficiency [9]. The degree of introduction e was 0.34, and the degree of recompression γ was 0.27.

3. Plant assessment and general operating parameters

A first calculation on the overall efficiency and the delivered power was performed by considering the ideal cycle depicted in Fig. 1.

Several steam inlet pressures and different plant arrangements were taken into account but the same baseline engine was considered in all the cases, that is the one used also in the experimental part of this work. Moreover both saturated and superheated Rankine cycles were evaluated as well as the technique of steam regeneration (Figs. 3 and 4).

Delivered work per cycle was calculated applying $\int p\,dv$ to the various phases as suggested in literature [9]. The displacement of the engine was 330 cm^3, its rotating speed was 3000 rpm and a volumetric efficiency of 0.7 was considered.

The results of the calculation are depicted in Figs. 5 and 6. It may be noted that the delivered power may range from 5 to 50 kW as envisaged in the introduction, depending mainly upon the steam inlet pressure. At the same time, thermal efficiency is

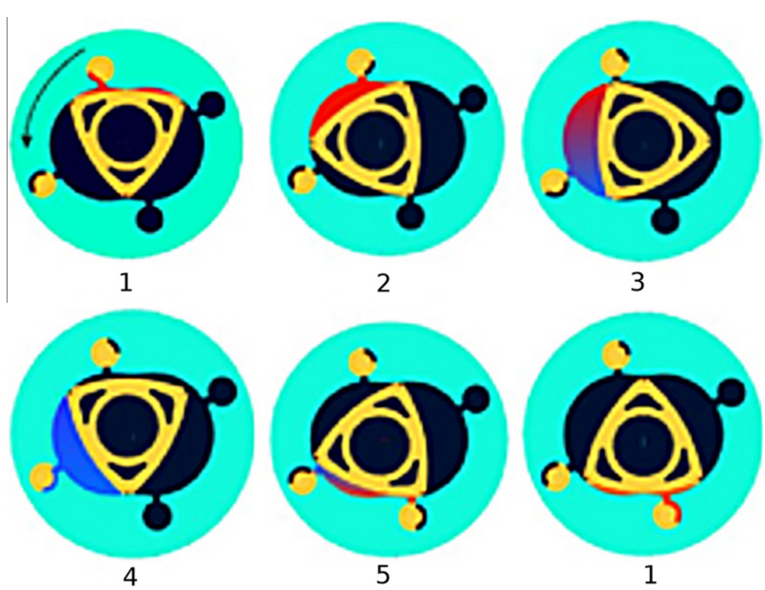

Fig. 2. Volumetric steam engine cycle phases.

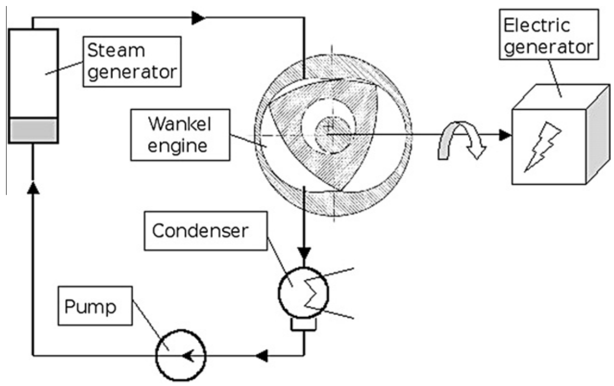

Fig. 3. Schematic layout of a plant using a steam Wankel engine.

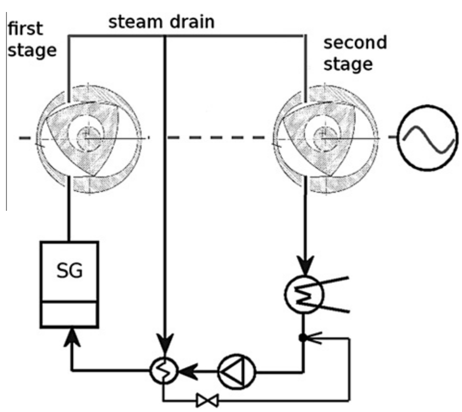

Fig. 4. Layout of a two-stage, regenerative steam Wankel engine plant.

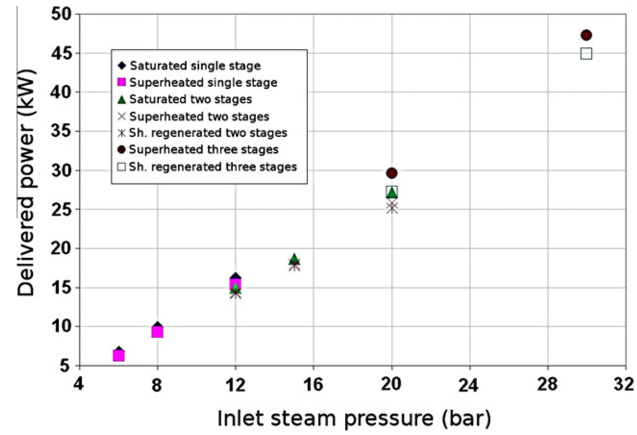

Fig. 5. Graphical representation of the results of cycle calculations in terms of delivered power as a function of the inlet steam pressure.

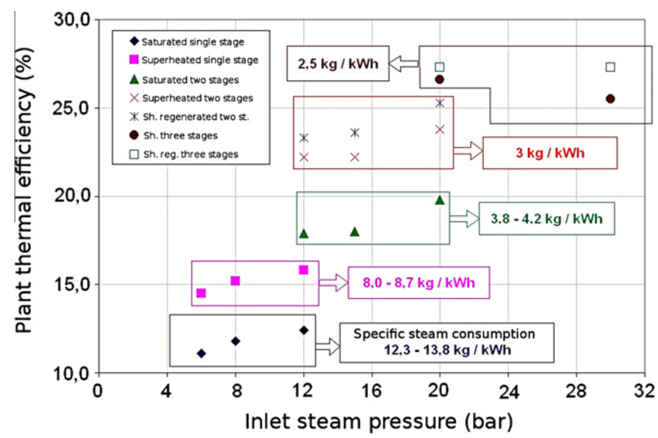

Fig. 6. Results of cycle calculations in terms of efficiency and steam specific consumption as a function of the inlet steam pressure.

deeply affected also by the plant asset. In facts if we turn from single to double staged expansion, a significant increase in thermal efficiency and into a strong reduction in steam specific consumption, up to 25%, may be attained.

4. Engine cycle modeling and analysis

The fluid dynamic behavior of the engine was analyzed through a numerical analysis that was supported by the experimental data that were used to validate the models developed within the work here presented. More in detail, a lumped parameter model was created within the AMESim simulation environment, while the experimental activity was carried out by testing the engine at the dyno (Fig. 7) and using the compressed air provided by a compressor as operating fluid at a pressure between 5 and 9 absolute bar, in a very similar way to what may be found in the literature [20]. Exhaust happened at atmospheric pressure.

The numerical model is composed by the three variable volume chambers with the proper phase shift between one and the other, and it takes into account the discharge losses through the valves. In this first work mechanical friction was taken into account by simply considering a fixed mechanical efficiency, equal to 0.9. This very simple assumption was made since the main address of this work was the analysis of the fluid dynamic phenomena.

The model of each of the three chambers of the engine is depicted in Fig. 8 where the intake and the exhaust valves are numbered as 1 and 2. Two valves are used both for the intake and the exhaust since every chamber completes two cycles per revolution. The operating chamber submodel (numbered as 3) accounts for heat transfer between the fluid and the walls by means of a proper exponent n of the polytropic law. The piston model (3) accounts for

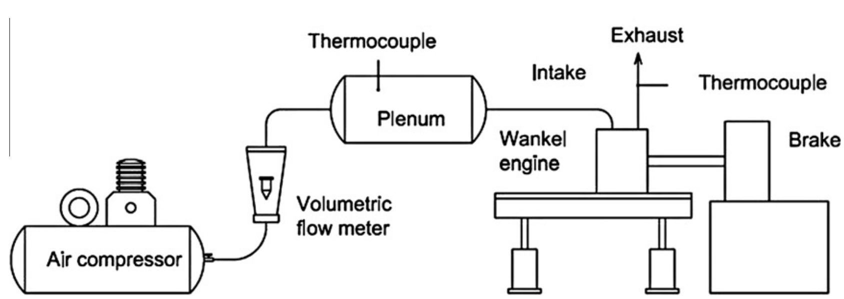

Fig. 7. Experimental apparatus used to test the Wankel prototype with compressed air.

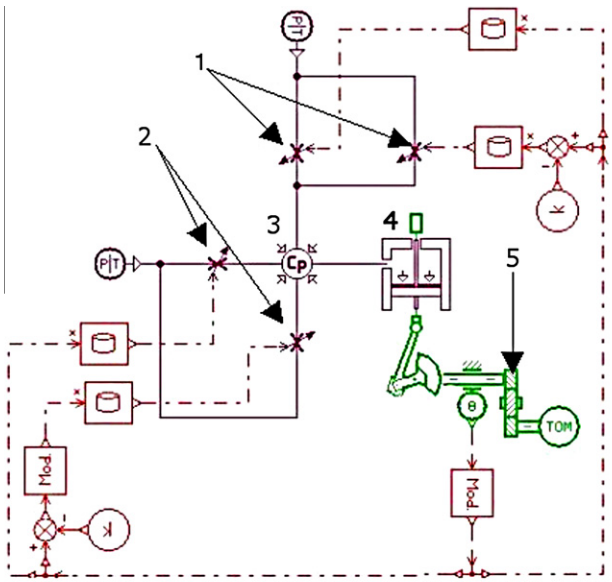

Fig. 8. Graphical representation of the numerical model used to simulate the fluid-dynamic behaviour of the steam Wankel engine.

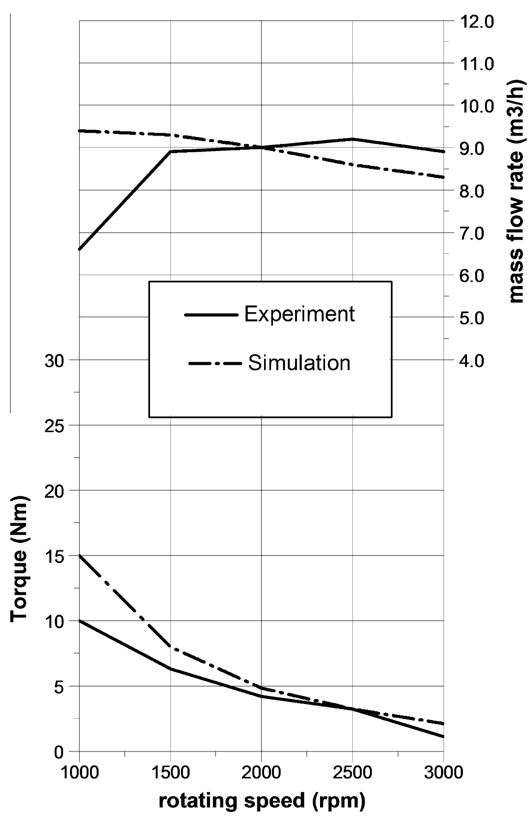

Fig. 9. Comparison between numerical analysis and experiments in terms of delivered torque and volumetric air flow rate-inlet air pressure equal to 6 bar.

the variation of volume, which is imposed by the model of the crank (4). This model was modified respect to the original crank model of the AMESim library since the kinematic of the Wankel engine is different from the conventional reciprocating engine. The gear model (5) takes into account that the rotor spins three times slower than the shaft.

As shown in Figs. 9 and 10, quite a good agreement was found between numerical and experimental data in terms of delivered torque and mass flow rate. It may be noted that the air mass flow

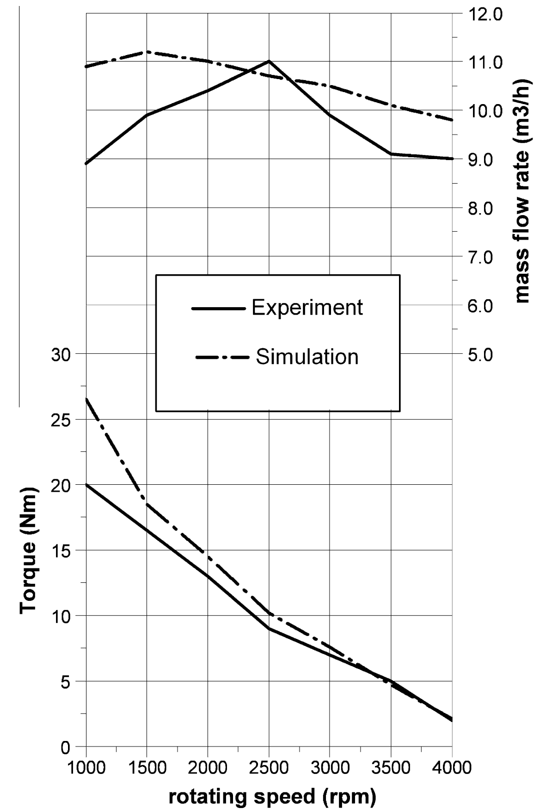

Fig. 10. Comparison between numerical analysis and experiments in terms of delivered torque and volumetric air flow rate - inlet air pressure equal to 9 bar.

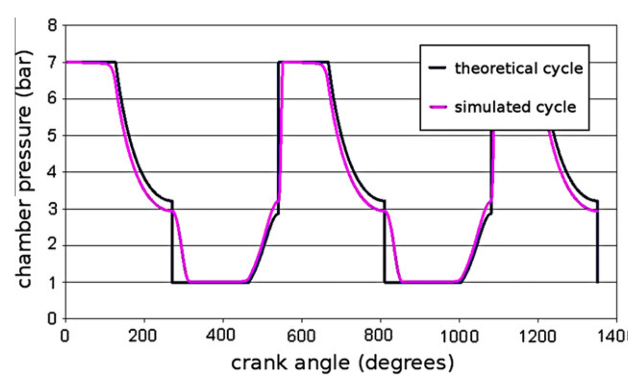

Fig. 11. Comparison between numerical analysis and experiments in terms of indicated pressure at 100 rpm.

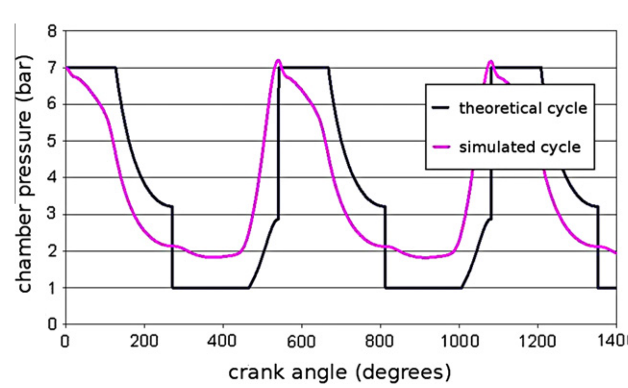

Fig. 12. Comparison between numerical analysis and experiments in terms of indicated pressure at 1000 rpm.

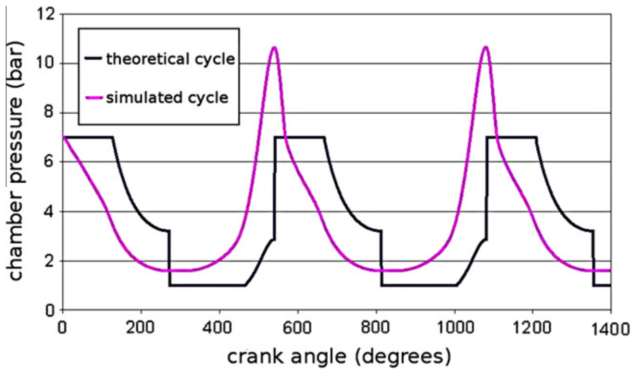

Fig. 13. Comparison between numerical analysis and experiments in terms of indicated pressure at 2500 rpm.

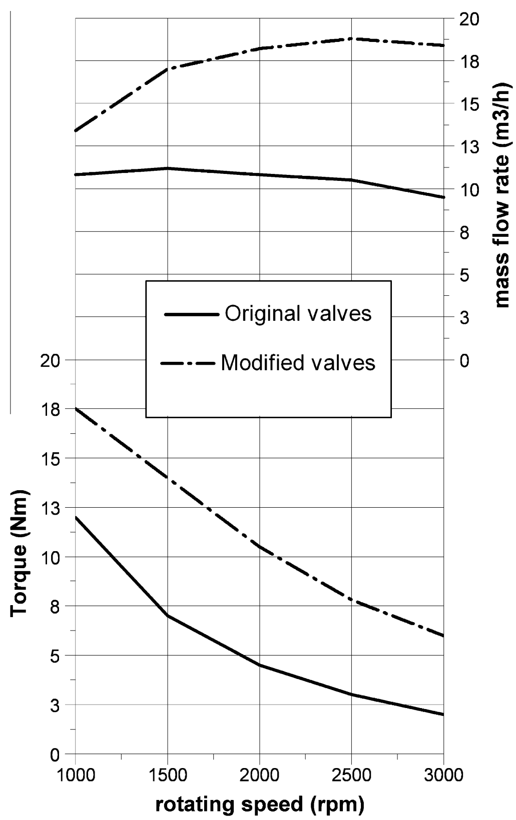

Fig. 14. Comparison in terms of delivered torque and volumetric air flow rate between the existing valves and the modified ones - inlet air pressure equal to 9 bar.

rate behavior is far from being linear with the rotating speed, and in order to explain this behavior the in-chamber pressure was analyzed.

The simulated pressure cycles were compared with the same ideal cycle, plotted by means of the equations suggested by the literature [9] at several rotating speeds, as reported in Figs. 11–13. At 100 rpm (Fig. 11) the engine pressure behavior is very close to the ideal one and only very slight differences can be noted, but if we increase the rotating speed, for instance, up to 1000 (Fig. 12) and then to 2500 rpm (Fig. 13), an increasing pressure drop across the exhaust valve may be observed.

This fluid dynamic loss that prevents all the air to be completely discharged and during the following recompression phase the remaining air pressure even exceeds the intake manifold pressure and steam backflow happens, thus reducing the volumetric efficiency.

The reduction of the recompression degree would provide a certain improvement in engine performances at the highest rotating speeds, but also a deterioration in efficiency at the lowest ones. The effect of the increase of the fluid dynamic permeability of the valves was then analyzed. A new valve design was conceived both for intake and exhaust valves. The design of this valve was conceived to increase the permeability of the cross section without a significant increase in bulk and weight.

A simulation was carried out in order to compare delivered torque and air mass flow rate when adopting an increased cross sectional area valve. Fig. 14 shows the resulting substantial increase in torque and mass flow rate using these new valves without modifications of the introduction and the recompression degrees.

5. Conclusions

Volumetric engines use may be considered for the distributed micro-generation in small size plants (5–50 kW) retaining at the same time an acceptable thermal efficiency (25%).

A numerical model was developed by means of the AMESim code, whose results were quite close to the experimental measurements. This model proved to be a very useful tool for the assessment of the engine performances and the fluid dynamic design. The use of this model showed that high rotating speeds may be attained only if a large permeability of the intake and exhaust sectional areas would be ensured.

The new test facility was built in the new Center for the Study on Biomasses for Energy (CRIBE) of the University of Pisa at San Piero a Grado (PI), and the tests of the engine in the real operating conditions using water steam as well as the evaluation of the use of new materials for the gaskets and new lubrication techniques [21,22] are being carried out. In a future publication the results of this experimentation will be shown.

References

[1] Bisone L. Reti elettriche per la trasmissione di energia e di informazioni (PLC) Stato dell'arte e possibilita' di utilizzo, rapporto CESI. 01/12/2004.
[2] Wolfe P. The implications of an increasingly decentralised energy system. Energy Policy 2008;36:4509–13.
[3] Lakew AA, Bolland O. Working fluids for low-temperature heat source. Appl Therm Eng 2010;30:1262–8.
[4] Chen H, Goswami Y, Stefanakos EK. A review of thermodynamic cycles and working fluids for the conversion of low-grade heat. Renew Sustain Energy Rev 2010;14:3059–67.
[5] Chen H, Goswami Y, Rahman MM, Stefanakos EK. A supercritical Rankine cycle using zeotropic mixture working fluids for the conversion of low-grade heat into power. Energy 2011;36:549–55.
[6] Acton O. Turbomacchine, ed. UTET; 1987.
[7] Guo T, Wang HX, Zhang S. Comparative analysis of CO_2-based transcritical Rankine cycle and HFC245fa-based subcritical organic Rankine cycle using low-temperature geothermal source. Sci China Technol Sci 2010;53(6):1638–46.
[8] Nguyen TQ, Slawnwhite JD, Goni Boulama K. Power generation from residual industrial heat. Energy Convers Manage 2010;51:2220–9.
[9] Caputo C, Acton O, Beccari A. Compressori ed espansori volumetrici. ed UTET; 1992.
[10] Quaggiotti V. Il motore Wankel, ed. Patron; 1972.
[11] Andale RF. The NSU-Wankel rotary engine. Automobile Engineer; 1960.
[12] Rotary engine design; 1989. ISBN:0-89883-695-6.
[13] Quaggiotti V. Le tenute dei motori volumetrici rotativi. L'Ingegnere, n.11; 1965.
[14] Shimizu R, Tadokoro T, Nakanishi T, Funamoto J. Mazda 4-rotor rotary engine for the Le Mans 24-hour endurance race. SAE paper 920309; 2009.
[15] Cole DE, Jones C. Reduction of emissions from the Curtiss–Wright rotary combustion engine with an exhaust reactor. SAE paper 700074; 1974.
[16] Mathias JA, Johnston J, Cao J, Priedeman DK, Christensen RN. Experimental testing of Gerotor and Scroll expanders used in, and energetic and exergetic modeling of, an organic Rankine cycle. J Energy Resour Technol – Trans ASME 2009;1(1).
[17] Badr O, Naik S, O'Callaghan PW, Probert SD. Expansion machine for a low power-output steam Rankine-cycle engine. Appl Energy 1991;39(2):93–116.

[18] Badr O, Naik S, O'Callaghan PW, Probert SD. Rotary Wankel engines as expansion devices in steam Rankine-cycle engines. Appl Energy 1991;39(1): 59–76.
[19] Badr O, Naik S, O'Callaghan PW, Probert SD. Wankel engines as steam expanders: design considerations. Appl Energy 1991;40(2):157–70.
[20] Guangbin L, Yuanyang Z, Liansheng L, Pengchen S. Simulation and experiment research on wide ranging working process of scroll expander driven by compressed air. Appl Therm Eng 2010;30:2073–9.
[21] Okularczyk W. Experimental investigations of guide rings made of UHMWPE and PTFE composite in water hydraulic systems. In: Archives of civil and mechanical engineering; 2004.
[22] Jia JH, Zhou HD, Gao SQ, Chen JM. A comparative investigation of the friction and wear behavior of polyimide composites under dry sliding and water-lubricated condition. Wear 2006;261(11–12):1293–7.